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Brief Papers

Cascaded sliding mode force control for a single-rod electrohydraulic actuator

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ABSTRACT

This paper investigates the force control problem of a single-rod electrohydraulic actuator system based on sliding mode strategy. On the basis of the force tracking error dynamics, in order to facilitate the controller design, the whole system is divided into a linear subsystem and a nonlinear subsystem. By forcing the output of nonlinear subsystem to track the expected fictitious input of linear subsystem, and specifying suitable sliding mode functions for nonlinear subsystem and linear subsystem respectively, the cascaded sliding mode controller is created according to the reaching law approach. The stability of the closed-loop system is proved. Simulation results verify the effectiveness of the proposed cascaded sliding mode force control method for the single-rod electrohydraulic actuator system.

1. Introduction

Electrohydraulic systems play an important role in wide variety of industrial and military applications, such as marine missions, oil and gas surveys, telerobotic operations, and thrust vector systems [1–5]. It is primarily due to their high durability, high power-to-weight ratios and rapid responses [1]. In recent years, various control approaches have been involved in electrohydraulic systems, such as model predictive control [4], sliding mode control [6], quantitative feedback control [7], fuzzy control [8], and backstepping control [9].

Applications interacting with the environment might require force control. To realize the desired force of hydraulic actuators is an important component in applications such as robotics, vibration isolation, and active suspensions [7]. Wang et al. [2] investigated the force control of a single-rod electrohydraulic actuator by using a feedback domination approach. Fuzzy logic control is applied to the electrohydraulic servosystem in [8]. A backstepping approach is used in [9] to design a nonlinear controller for force control of a single-rod electrohydraulic actuator.

The purpose of this paper is to construct a brief and effective controller for a single-rod electrohydraulic actuator system with desired force trajectory. In order to make the design of force feedback control system be convenient, the system is divided into a linear subsystem and a nonlinear subsystem. Because of the good performance provided by sliding mode strategy, by forcing the output of nonlinear subsystem to track the desired fictitious input of linear subsystem, a cascaded sliding mode controller is constructed. After these efforts, the presented force feedback control method is brief and the practical applicability can be increased. The stability of the closed-loop system is proved based on Lyapunov theory.

The remainder of this paper is organized as follows: in Section 2, the considered single-rod electrohydraulic system is given. In Section 3, by taking force as a state variable, the dynamics is deduced at first; and then the whole system is divided into a linear subsystem and a nonlinear subsystem; next the cascaded sliding mode controller is created for the system and the stability of the system is proved. Simulation on a single-rod electrohydraulic system is illustrated in Section 4. Section 5 draws the conclusions of the paper.

2. Preliminary

Consider the single-rod electrohydraulic system depicted in Fig. 1. The actuator's piston dynamics is

$$m\ddot{y}_L = F_{hydr} - c\dot{y}_L - F - F_f \tag{1}$$

where y_L, $\dot{y}_L$, $\ddot{y}_L$ are the displacement, speed and acceleration of the actuator's piston, respectively. m is the mass of the servovalve and c represents the equivalent viscous damping coefficient. F is the load's force which is applied to the end of the piston rod. The force is measured by the force sensor, and the force signal is fed back to the computer through the interface board. F_{hydr} represents the driving

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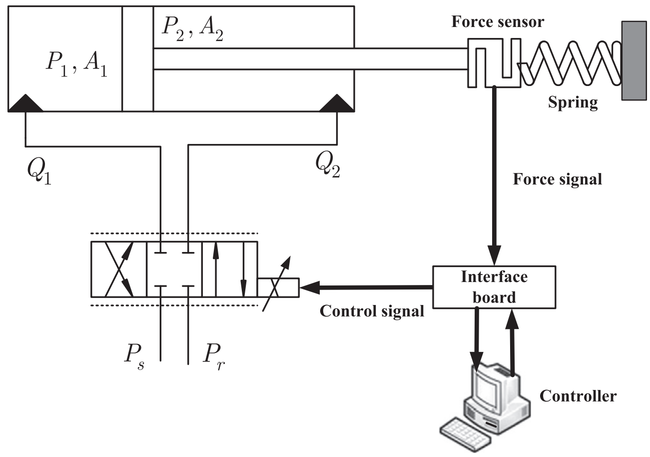

Fig. 1. Electrohydraulic servo system [2]

force generated from the electrohydraulic actuator and acting on the piston. F_f is the friction force, and assumed can be ignored.

The driving force F_{hydr} can be described by

$$F_{hydr} = P_1A_1 - P_2A_2$$

where P_1 and P_2 are cylinder pressures of the extend and retract chambers, respectively. A_1 and A_2 are the areas of the extend and retract chambers.

Neglect all leakage flow components, the dynamics of cylinder pressures can be written as

$$\dot{P}_1 = \frac{\beta_e}{V_1}(Q_1 - A_1\dot{y}_L)$$
$$\dot{P}_2 = \frac{\beta_e}{V_2}(A_2\dot{y}_L - Q_2) \tag{2}$$

where $V_1 = V_{10} + A_1y_L$ and $V_2 = V_{20} - A_2y_L$ are the volumes of the extend and retract chambers, V_{10} and V_{20} are corresponding volumes when y_L=0, respectively, β_e is the effective bulk modulus of the hydraulic oil, and Q_1 and Q_2 are the flow rates of the chambers.

Consider that the valve has a linear flow gain characteristic [12], the flow rates Q_1 and Q_2 can be related to the voltage input V_{in} directly by a linear mapping as

$$Q_1 = \begin{cases} KV_{in} & \text{if } \dot{y}_L \geq 0 \\ \gamma KV_{in} & \text{if } \dot{y}_L < 0 \end{cases}$$
$$Q_2 = \begin{cases} \dfrac{K}{\gamma}V_{in} & \text{if } \dot{y}_L \geq 0 \\ \gamma KV_{in} & \text{if } \dot{y}_L < 0 \end{cases} \tag{3}$$

where K is the flow/signal gain of the valve and γ is the flow factor of single-rod cylinders.

Suppose the load is simply modelled as a pure spring, neglect the mass and stiffness of the force sensor, then it yields

$$F = k_sy_L$$

where $k_s > 0$.

Thus

$$y_L = \frac{F}{k_s}$$
$$\dot{y}_L = \frac{\dot{F}}{k_s}$$
$$\ddot{y}_L = \frac{\ddot{F}}{k_s} \tag{4}$$

Substituting (2)–(4) into (1) yields

$$\ddot{F} = \frac{k_s}{m}F_{hydr} - \frac{c}{m}\dot{F} - \frac{k_s}{m}F \tag{5}$$

By defining the state variables as $x_1 = F$, $x_2 = \dot{F}$, $x_3 = \dot{F}_{hydr}$ and $x = [x_1, x_2, x_3]^{\mathrm{T}}$, the dynamic model can be written as

$$\begin{cases} \dot{x}_1 = x_2 \\ \dot{x}_2 = -\dfrac{k_s}{m}x_1 - \dfrac{c}{m}x_2 + \dfrac{k_s}{m}x_3 \\ \dot{x}_3 = g_2u - g_1x_2 \\ y = x_1 \end{cases} \tag{6}$$

where

$$g_1 = \frac{\beta_eA_1^2}{k_sV_{10} + A_1x_1} + \frac{\beta_eA_2^2}{k_sV_{20} - A_2x_1}$$
$$g_2 = \frac{\beta_eA_1Kk_s}{k_sV_{10} + A_1x_1} + \frac{\beta_eA_2Kk_s}{(k_sV_{20} - A_2x_1)\gamma}$$

3. Cascaded sliding mode controller design

Let F_r and $\dot{F}_r$ be the force and force-rate desired signals, respectively. F_{hydr_r} is the desired signal for F_{hydr}. Define $X = [x_1, x_2]^{\mathrm{T}}$, $\xi = x_3$, $X_d = [F_r, \dot{F}_r]^{\mathrm{T}}$, and the tracking errors of X and F_{hydr} are $e_x = X - X_d = [e_{x_1}, e_{x_2}]^{\mathrm{T}}$ and $e_h = \xi - F_{hydr}$. Suppose $\dot{F}_r = 0$, $F_{hydr} = 0$, $\dot{F}_{hydr_r} = 0$, then

$$e_x = \begin{bmatrix} x_1 - F_r \\ x_2 \end{bmatrix}$$
$$e_h = \xi$$

Hence

$$x_1 = e_{x_1} + F_r$$
$$x_2 = e_{x_2} \tag{7}$$

According to (6) and (7), the dynamic of e_x is

$$\begin{aligned} \dot{e}_x &= \dot{X} - \dot{X}_d \\ &= \begin{bmatrix} x_2 \\ -\dfrac{k_s}{m}x_1 - \dfrac{c}{m}x_2 + \dfrac{k_s}{m}x_3 \end{bmatrix} \\ &= \begin{bmatrix} e_{x_2} \\ -\dfrac{k_s}{m}(e_{x_1} + F_r) - \dfrac{c}{m}e_{x_2} + \dfrac{k_s}{m}\xi \end{bmatrix} \\ &= AX + b\xi \end{aligned} \tag{8}$$

where

$$A = \begin{bmatrix} 0 & 1 \\ -\dfrac{k_s}{m} & -\dfrac{c}{m} \end{bmatrix}, \quad b = \begin{bmatrix} 0 \\ \dfrac{k_s}{m} \end{bmatrix}$$

According to (6), there exists

$$\dot{\xi} = -g_1e_{x_2} + g_2u \tag{9}$$

Obviously, (8) is a linear subsystem while (9) is a nonlinear subsystem.

The procedure of obtaining cascaded sliding mode controller includes two steps. Firstly, ξ is viewed as a fictitious input for the linear subsystem (8), an auxiliary control ξ_d is designed to be the desired signal for ξ. Secondly, according to subsystem (9), control input u is determined such that the actual ξ tracks the desired ξ_d well.

According to sliding mode theory, a sliding surface with desired performance should be created first of all, and then a suitable control law is required to drive states to origin along with the sliding surface.

For the linear subsystem (8), the sliding function is defined as

$$s_x = \sigma e_x \tag{10}$$